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INFLUENCE OF DYNAMIC CHARACTERISTICS OF MACHINERY CONTAINING UNIVERSAL JOINTS ON THE CONSTRUCTION OF INDUSTRIAL PREMISES

Abstract. It is generally known that the universal joints are used in the mechanisms of many branches of modern industry, from mining to space. The relative simplicity of manufacture and low cost also contribute to the spread of these joints. However, during the using of mechanisms driveline vibrations arising in the premises are transmitted to the constructions of the building. This must be considered. The article examines the work of the double driveline in conjunction with the vibroimpact mechanism of the machine to clean the pipelines from the isolation. Being studied the influence of dynamic loads on the maximum equivalent stresses in the links of the cleaning machine. To exclude dangerous operation modes performed modal analysis and found own system frequency. Moreover, we attempt to optimize the most loaded part of system (driveline yoke) from the point of equal strength basing on the obtained results of calculations of the stress-strain state (hereinafter SSS). For this purpose an analysis of the influence of the geometric parameters of a yoke for strength is made and it established the most rational correlation of observed parameters. Thereby, the study of SSS, taking into account the dynamic factors, modal analysis, and development of recommendations for the design of equal strength eyelets of driveline are the main objectives of this work.

Keywords: driveline, dynamic coefficient, modeling of SSS with taking into account the dynamics, modal analysis, equal strength eyelet.

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ВЛИЯНИЕ ДИНАМИЧЕСКИХ ХАРАКТЕРИСТИК МАШИН И МЕХАНИЗМОВ С УНИВЕРСАЛЬНЫМИ ШАРНИРАМИ НА КОНСТРУКЦИИ ПРОИЗВОДСТВЕННЫХ ПОМЕЩЕНИЙ

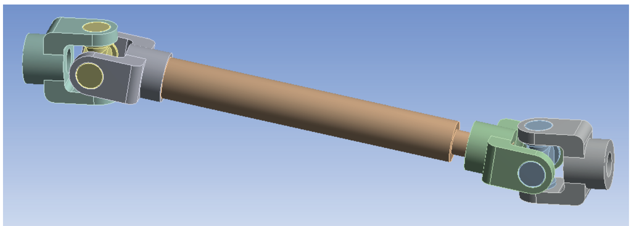
Аннотация. Общеизвестно, что универсальные шарниры применяются в механизмах многих отраслей современной промышленности: от горнодобывающей до космической. Относительная простота их изготовления и невысокая стоимость также способствуют распространению этих шарниров. Однако при эксплуатации механизмов с карданными передачами в помещениях возникающие колебания передаются на строительные конструкции. Это необходимо учитывать. В статье рассматривается работа сдвоенной карданной передачи в связке с виброударным механизмом машины для очистки от изоляции магистральных трубопроводов. Изучается вопрос влияния динамических нагрузок на максимальные эквивалентные напряжения, возникающие в звеньях очистной машины. Для исключения опасных режимов работы проводится модальный анализ и выявляются собственные частоты системы. Кроме того, опираясь на полученные результаты расчетов напряженно-деформированного состояния (далее НДС) делается попытка оптимизации наиболее нагруженного узла системы (вилки) с точки зрения равнопрочности. Для этого проводится анализ влияния геометрических параметров вилки на прочность и устанавливается наиболее рациональное соотношение рассмотренных параметров. Таким образом, исследование НДС с учетом динамических факторов, модальный анализ и разработка рекомендаций по конструированию равнопрочных проушин карданных передач являются основными задачами данной работы.

Ключевые слова: карданная передача, динамический коэффициент, моделирование НДС с учетом динамики, модальный анализ, равнопрочность проушин.

1. Introduction

At the present time machinery with universal joints are used in many industries: automotive, petroleum, mining, aerospace, agriculture, etc. Moreover, these mechanisms find their use in the production of other products: in metal rolling mills, in machines for the removal of the insulating coating of pipelines, in the robot manipulator and others. Similar equipment is installed i n industrial premises; vibrations are transmitted from it to the building construction, and thus should be considered when designing structures. The authors studied the technology mechanism, in which composition is used driveline and vibroimpact tool (Fig. 1) [3].

a



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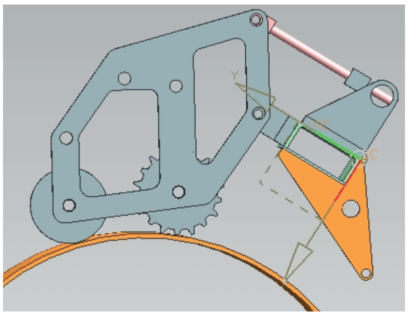


Fig. 1. Driveline (a) and vibroimpact fragment of mechanism (b)

Analysis of transmission dynamics is presented in the works [1], [2]. The mathematical model, made on the basis of the principle of D'Alembert, was the subjected to consideration:

$$I_1\ddot{\varphi}_1 + S_{12}(\varphi_1-\varphi_2) + K_{12}(\dot{\varphi}_1-\dot{\varphi}_2) = M_1$$

$$I_2\ddot{\varphi}_2 - S_{12}(\varphi_1-\varphi_2) - K_{12}(\dot{\varphi}_1-\dot{\varphi}_2) + S_{23}(\varphi_2-\varphi_3)F_1 = -M_{fr,1}$$

$$I_3\ddot{\varphi}_3 - S_{23}(\varphi_2-\varphi_3) + S_{34}(\varphi_3-\varphi_4) + K_{34}(\dot{\varphi}_3-\dot{\varphi}_4) = 0$$

$$I_4\ddot{\varphi}_4 - S_{34}(\varphi_3-\varphi_4) - K_{34}(\dot{\varphi}_3-\dot{\varphi}_4) + S_{45}(\varphi_4-\varphi_5)F_2 = -M_{fr,2}$$

$$I_5\ddot{\varphi}_5 - S_{45}(\varphi_4-\varphi_5) + S_{56}(\varphi_5-\varphi_6) + K_{56}(\dot{\varphi}_5-\dot{\varphi}_6) = 0$$

$$I_6\ddot{\varphi}_6 - S_{56}(\varphi_5-\varphi_6) - K_{56}(\dot{\varphi}_5-\dot{\varphi}_6) = -M_t\ .$$

A dot placed over the function name represents a time derivative; F_1 and F_2 — kinematic functions, which expressed nonlinear kinematic relationship between the links; M_1 и M_t — moments on the driving and driven links; M_{fr} — moments of friction in the joints; ω — angular velocity; φ_i — angles of rotations of inertial mass; S_{ij} — reduced rigidity of the links; I_i — reduced moment of inertia of mass; K_{ij} — coefficient of viscoelastic damping [2].

The solution of the system was carried out using a computer program developed by the authors. As a result, dynamic moments in the links of the hinge system are defined and dynamic coefficients calculated — Kd, which in this article are used to calculate the refined yoke SSS.

2. Research methods and results

Refined equivalent stresses in the yoke.

Refined equivalent stresses from the static and dynamic loads are defined as follows:

$$\sigma_{\text{dyn}} = \sigma \cdot \text{Kd}, \qquad (2)$$

where σ — equivalent stresses obtained from the SSS formula solving.

In this work during the study of yoke strength contact interaction between the transmission elements, handling loads and dynamic loads are considered (included in equation (1)).

For the hinge system modeling of SSS was carried using the finite element method. The solid model of the hinge with a specified application load is shown in Fig. 2. It also shows the subdivision of finite element model of an irregular grid, consisting of volume elements in the form of a hexahedron and tetrahedron. The calculations

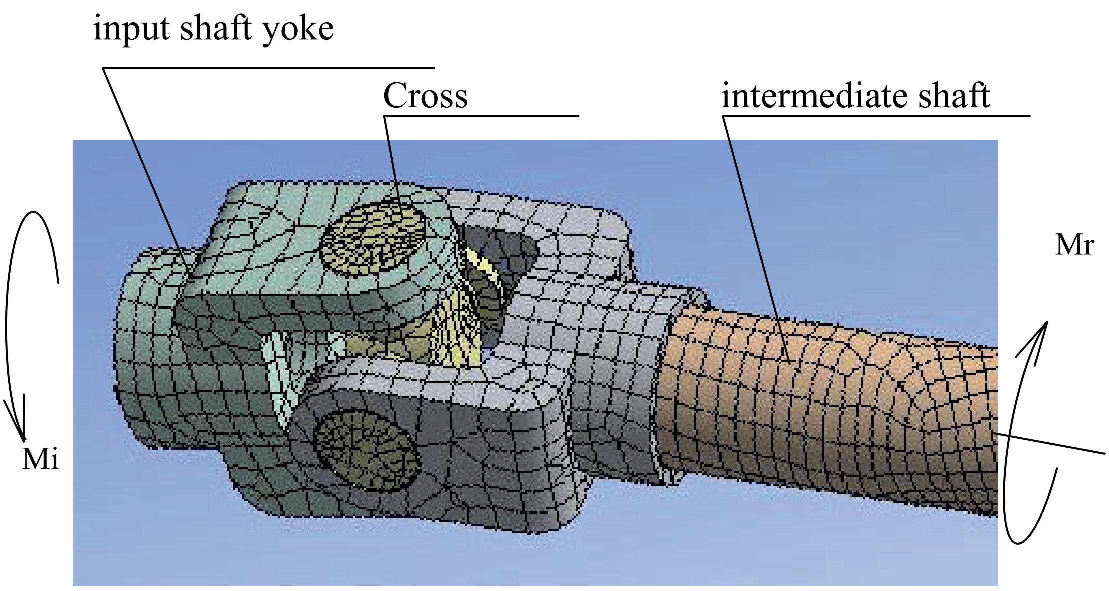


Fig. 2. Finite-element model of the universal joint, where the Mi — torque at the input, Mr — moment of resistance

used the bilinear diagram of deformation of the material. Contact interaction were simulated using the method of "Updated Lagrangian" antifriction bearing friction coefficient is assumed to be 0.005.

In the simulation, accepted the assumption that the friction surface in the bearing assembly is averaged surface passing through the axis of each of the bearing rollers.

SSS calculation was performed for a range of angular positions of the input shaft yoke of driveline with an interval angle of 5 degrees. Fig. 3 shows the most loaded area of the driveline, which occur in the eyelets of yokes. The calculation results are visualized and constitute a stress field in the form of areas with a different shade of gray. The maximum equivalent stress in the eyelet yokes marked the darkest color. And, depending on the angle of rotation of yoke, these stress zones redistributed. According to the stress field intensity distribution it can be concluded that in order to achieve equal strength necessary to change its yoke geometric parameters, combined with an increase in compliance elements all across the driveline.

Optimization of yoke geometry

This work is an attempt to optimize the geometric shape of yoke of driveline from the point of equal strength. For this purpose was made a series of SSS calculations of the universal joint (Fig. 1) with modified geometric parameters R, L, H (Fig. 4). Fig. 4 shows the yokes SSS with different shape from the yoke shown in Fig. 3. In this Fig. 4a shows a SSS eyelets reinforced by adding material at the perimeter; Fig. 4b — enhanced by the addition of the material in the upper and lower sectors of the eye; Fig. 4c, in — enhanced by the addition of material around the eyelet holes.

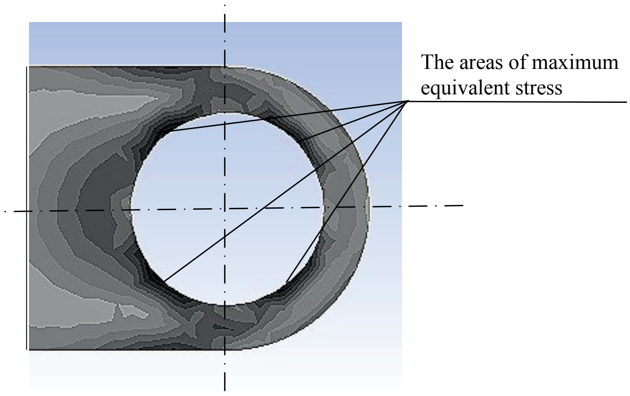


Fig. 3. Distribution of equivalent stress in the eyelet of yoke of driveline

The analysis of SSS from Fig. 4 shows that in case of "a" addition of material on an external contour of eyelet reduces tension throughout the part, but it does not lead to uniform strength; in the case of "b" is a reduction of maximum stress in the right inside of the eyelet; in the case of "c" a reduction in the overall maximum stress, as well as much more equal stress distribution throughout the eyelet relative to the previous embodiments.

Basing on the results, we take the form of eyelet option "c" as recommended in the design. Next, we investigate the effect of geometrical parameters R, L and H (Fig. 4c)

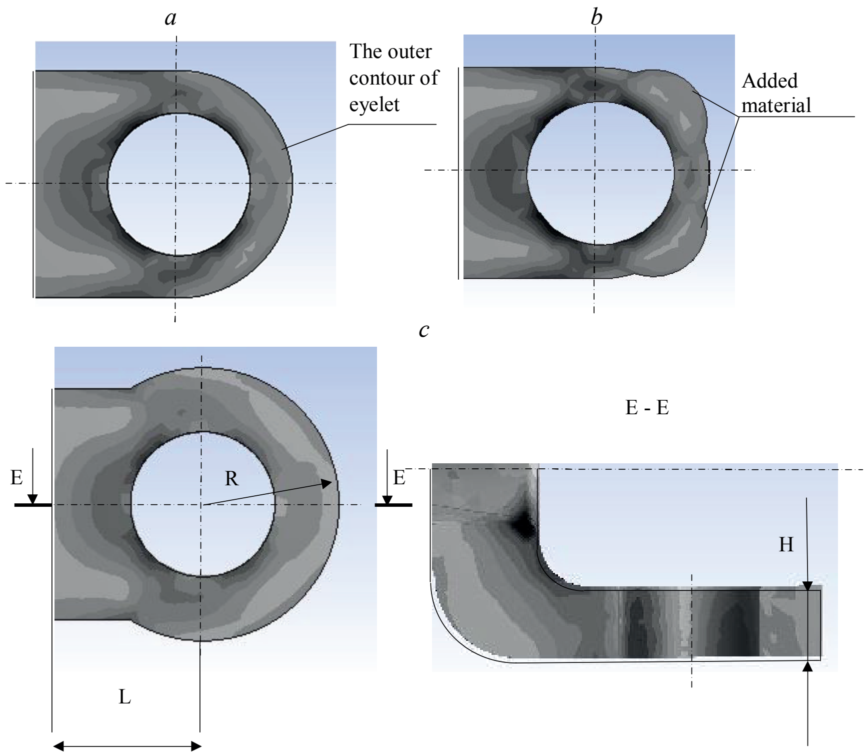


Fig. 4. The equivalent stress distribution in the yokes of different geometric shapes

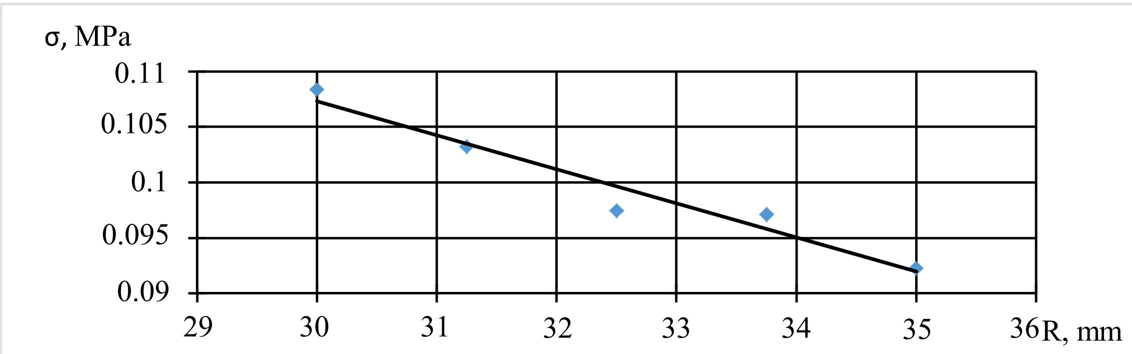
on the strength of the eyelets, which perform calculations sequentially changing each parameter. Unit torque is accepted as load (M = 1 N/m), the initial values of the parameters: R = 30 mm, L = 36 mm, H = 17 mm. Results are presented as graphs in Figs. 5. The graph "a" shows that the dependence of the maximum equivalent stress on the external eyelet radius R is almost linear — with an increase in the range stress decreases. The dependence of the maximum equivalent stress on the distance L between the axis and the plane of securing eyelets on the "b" chart is more complicated, but in general there is a trend of increasing stress with increasing distance. And the graph "c" indicates that the maximum equivalent stresses decrease with increasing thickness of the eyelet H.

Consequently, to achieve a equal strength design eyelets, investigator must increase the values of the parameters R and H, and reduce the value of the parameter L as long as the system will meet the other specified requirements (weight, dimensions, etc...). Furthermore, should be avoided stress concentrators, which are located in areas of sharp changes in geometry.

Modal analysis

Finding the natural frequencies is an integral part of the designing of any mechanical system undergoing periodic stress. Thus, coincidence units driveline speeds with natural frequency can result resonances in the system, therefore, to higher loadings, premature mechanical failure of the system and non-proper performance of the functions performed by the system. Conducted by a program modal analysis of spatial driveline (Fig. 5) revealed that the lowest resonance

a



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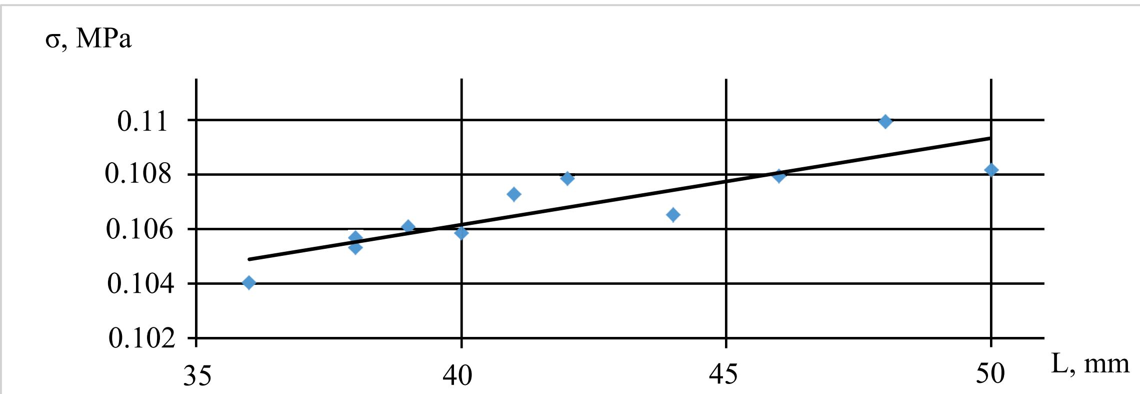

c

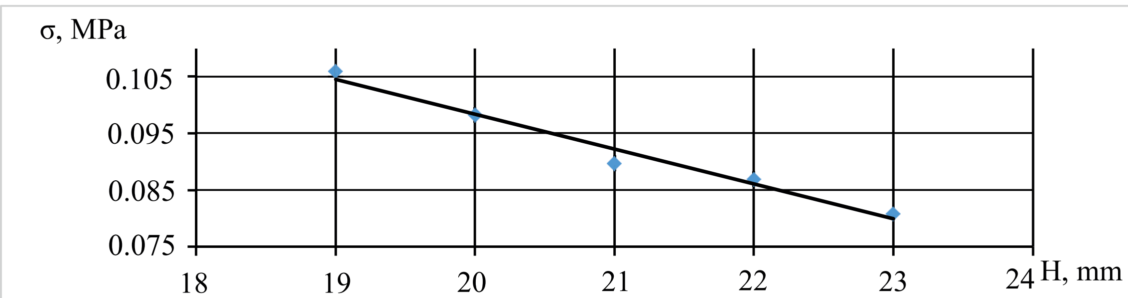

Fig. 5. Charts change the maximum equivalent stress σ depending eyelets parameters R, L, H

frequency is 392.9 Hz — it is by an order above the system operating range of angular velocities (0–33,3 r/sec). That means, considered system operates in preresonance area to the left of "resonace zone", which indicates the correct choice of the geometric parameters of the system. In addition, the natural frequencies of structures that use mechanisms that have the dynamics should be taken into consideration, in order to avoid resonance effects on a larger scale.

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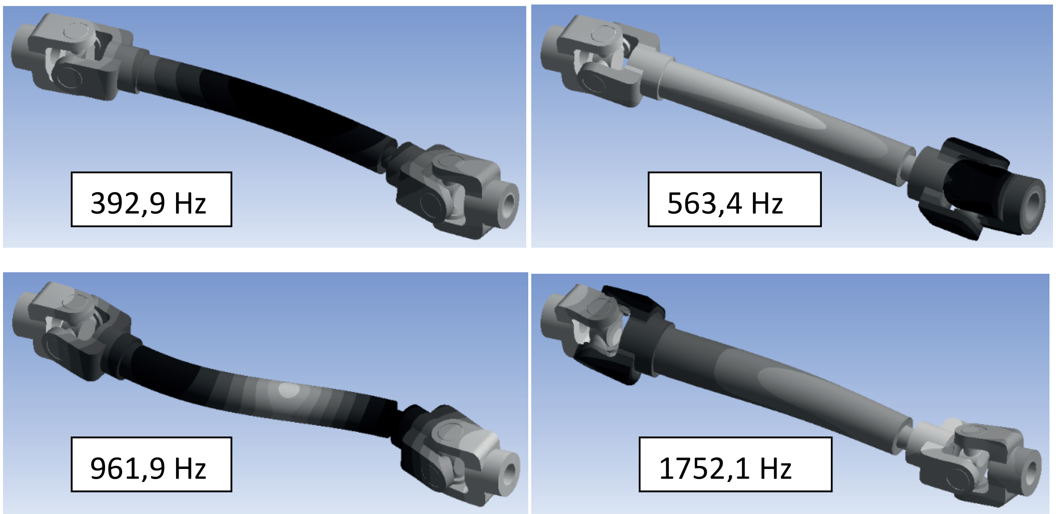

Fig. 6. The resonance frequency of natural oscillations of the hinge system

3. Conclusion

As a final point, the results of the work it turned out that the areas of maximum stress are on the inner contours of eyelets, and these areas are shifted according to the change of position of normal pressure forces and the geometric parameters of the system. It was observed that the shape of eyelets, close to that shown in Fig. 3 by the letter "c", is more equal strength in comparison with other cases considered. For this option is studied the influence of the major geometric parameters of eyelets on the distribution of equivalent stress, determined laws and given recommendations for the considered variation of parameters. Furthermore, the example above shows the need of modal analysis of the hinge systems to exclude operation of the system in the resonance modes.

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